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FAILURE ANALYSIS OF COMPRESSION HELICAL SPRING USED IN THE SUSPENSION SYSTEM BY FEA

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ABSTRACT

This paper is an investigation and comparison between ASTM A227 and Chromium vanadium AISI 6150 as a material in springs used in vehicles suspension systems. Numerical analysis using ANSYS 16.1 software to calculate the early failure and predict the resistance of the material and under the effect of the harmonic forces of 4250 N and 4500 N and the frequency of 100 Hz at the rate of increase 5 Hz. The results have been shown is concluded, that by using Chromium vanadium AISI 6150 would be highly durable and more efficient for use in suspension systems in the vehicles through riding on the slopes or bumped road. Also, can choose the appropriate material according to loads or weight of the vehicles and operating conditions.

KEYWORDS: FEA, Helical Compression Spring, Shear Stress, Deformation & Frequency Response

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Original Article

1. INTRODUCTION

Compression helical springs are important elements in the mechanical design. It's known that the spring function is the absorption of shocks and vibrations, also absorb of energy, therefore, it is necessary to study of the failure of springs, which explained in the shear stress and deformation effects of strength it, under the static or variable loads. In this study, the focus was placed on the resistance of the springs under variable loads (fatigue failure effect). The previous studies have emerged as the following. [1] Studied Max. shear stress and gets percentage error about (1.5%–4 %) as a compared numerical analysis with theoretical calculations. In addition, interested in ASTM A227 as the material used in vehicles of three-wheels. [2] Reviewed the studies failure of helical springs under the wiggle loading and interested in the calculation of fatigue and max. shear stresses and deflection distribution by FEM programs had been using of performance mesh simulation. In addition, the comparison of the theoretical results with numerical results by FEA, it will conceivable to help designers for the design of springs to prevent failure. [3] Studied distribution of stress, characteristic of materials, manufacture, and failure. Moreover, obtained max. deflection is (0.313807 mm in the y-direction), also a high difference in stress between theoretical and FEA (where 1267 MPa in FEA and 1010 MPa in theoretical). In addition, decided the spring failure is possible occurs in the critical section is more than sections compared to others section. [4] Interested in the failure of steel grade 60Si2CrVA as a spring used in heavy-duty vehicles, and focused on a set of recommendations are important to select of non-closed ends for design to avoid of wear, and corrosion of springs. [5] Investigated the failure reduction of springs under the static and dynamic loading, used Pro-E to 3D solid modelling and ANSYS software to analyze and concluded his study to the strength of springs are influenced through the change of coil diameter to maximum expecting to gets the appropriate elastic and inappropriate for the conditions. [6,7] Reviewed to the discussed of mechanical springs design used in the automobiles are perfectly very necessary for designing and analysis which involving stress distribution and maximum extension and different of failure mode. Also, the springs subjected to the oscillating loads through the service, supplement, divers software design such as ANSYS, Solid Works, Pro-E, and

CATIA, etc., had been used for implementing the stress analysis. Nearly in all cases, fatigue stresses, shear stresses, and max. displacement calculations, played as a role considerable in the mechanical spring design. [8] Focused on two parts of the study (analytical by ANSYS 16 and experimental test), from FEA gets of the fatigue life (6.36 x 10^5) cycles and max. shear stress is (715.93 MPa) with percentage error (2.64 % and 2.61% at max. shear stress), so as for strain gauges in experimental with used five difference of cycles and obtained the genuine of fatigue life are estimated and targeted failures can be evaded, however, improved the aspect safe of the vehicles. [9] Interested in displacement and shear stress and used two materials (chromium-vanadium AISI 6150 and low-carbon steel AISI 1018), used ANSYS R14.5 to analytical with three forces different (3432.3, 4413, and 4903.3 N), and concluded that by using the Stainless Steel alloy in the springs, the resisting capacity of it will be highest and sufficient for the purpose of domestic. Through the literature studies, indicate that the material used is the basic element that plays a role in mechanical designs.

2. SELECTION OF MATERIALS

Generally, steel alloys are used in the suspension system for high resistance to shock, vibration, and variables loading. Therefore is necessary to select the appropriate material for each purpose. This study focused on two different materials (one is ASTM A227 and another is Chromium Vanadium AISI 6150) to investigate the resistance of each material under variable loads conditions. Table (1) shows the chemical composition and the table (2) shows the properties of materials. [10, 11, 12]

Table 1: Chemical Composition

Material	Chemical Composition							
	Fe %	Cr %	Mn %	C %	Si %	V %	S %	P %
ASTM A227	97.4–99.1	------	0.3–1.3	0.45–0.85	0.15–0.35	------	0–0.05	0–0.04
Chromium Vanadium AISI 6150	96.7–97.7	0.8–1.1	0.7–0.9	0.48–0.53	0.2–0.35	0.15	0–0.04	0–0.035

Table 2: Properties of Materials

Material	Properties							
	Brinell Hardness	Modulus of Elasticity GPa	Fatigue Strength MPa	Poisson's Ratio	Shear Modulus GPa	Ultimate Tensile Strength MPa	Yield Tensile Strength MPa	Density kg/m^3
ASTM A227	500–640	198.6	900–1160	0.29	80.7	1720–2220	1430–1850	7850
Chromium Vanadium AISI 6150	200–350	203.4	300–750	0.29	77.2	630–1200	420–1160	7830

3. COMPRESSION HELICAL SPRING CALCULATION

Table (3) shows the spring parameters, and figure (1) explained the standard of helical compression spring.[13]

Table 3: Spring Parameters

Parameters	Symbol	Units
Wire Diameter	d	*mm*
Mean Diameter	D	*mm*
Outer Diameter	D_o	*mm*
Spring Index	C	--------
Free Length	L_F	*mm*
Spring Rate	k	*N/mm*
Pitch	p	*mm*

Number of Active Turns	n	--------
Number of Total Turns	n'	--------
Deflection	δ	mm
Modulus of rigidity for the spring material	G	GPa
Shear Stress Factor	K_S	--------
Wahl's Stress Factor	K	--------
Maximum Shear Stress	τ_{max}	MPa
Mean Shear Stress	τ_m	MPa
Variable Shear Stress	τ_v	MPa
Axial Load	W	N
Mean Load	W_m	N
Variable Load	W_v	N

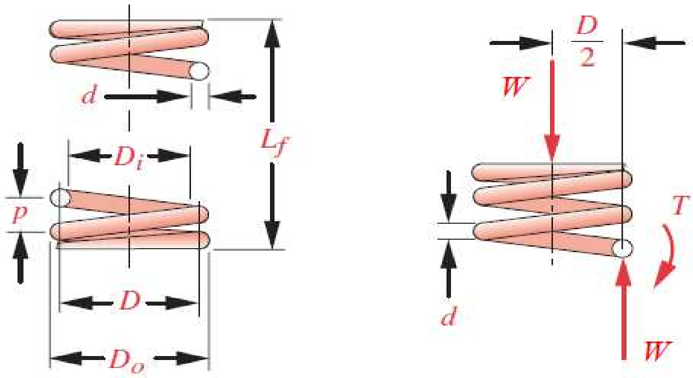

Figure 1: Compression Helical Spring.

3.1 Maximum Shear Stress

The helical compression springs are commonly subjected to two types of shear stresses. One is direct shear stress and the other is torsional shear stress. The equations (1–2) represent direct and torsional deflections, while the equation (3) represents the maximum shear stress.

Direct Shear Stress $$\tau_d = \frac{4 \times W}{\pi \times d^2} \tag{1}$$

Torsional Shear Stress $$\tau_T = \frac{8 \times W \times D}{\pi \times d^3} \tag{2}$$

Max. Shear Stress $\tau_{\max} = \tau_d + \tau_T$

$$\tau_{\max} = \frac{4 \times W}{\pi \times d^2} + \frac{8 \times W \times D}{\pi \times d^3}$$

$$\left.\begin{aligned} \tau_{\max} &= K_S \times \frac{8 \times W \times D}{\pi \times d^3} \\ \tau_{\max} &= K \times \frac{8 \times W \times D}{\pi \times d^3} \end{aligned}\right\} \tag{3}$$

Where: $K_S = 1 + \left(\frac{1}{2D}\right)$, and $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$

When the helical springs are subjected to fatigue loading are designed by used the *Soderberg line method* (1930). The spring materials are commonly examined for endurance torsional strength under repeated stresses that changes from zero to a maximum. The equations (4&5) represents mean shear and variable stresses.

$$\tau_m = K_S \times \frac{8 \times W_m \times D}{\pi \times d^3} \tag{4}$$

$$\tau_v = K \times \frac{8 \times W_v \times D}{\pi \times d^3} \tag{5}$$

Where: $W_m = \frac{W_{\max} + W_{\min}}{2}$, and $W_v = \frac{W_{\max} - W_{\min}}{2}$

3.2 Deflection of Compression Helical Spring

Deflection of helical springs are different according to the carry tensile or compression loading, there are required some means of transferring the loads from the support to the spring body, and the spring is subjects with initial tension. The equation (6) shows of the spring deflection.

$$\delta = \frac{8 \times W \times C^3 \times n}{G \times d} \tag{6}$$

4. FINITE ELEMENT ANALYSIS

Table (4) shows the geometry parameters, while the main purpose of using the finite element analysis is to reduce error through the reduction and simplification of equations, especially when it is in the angles of the helical as well as the modelling method. In addition, the accuracy of the results which can be obtained by creating of a three-dimensional geometry of solid model as shown in figure (2) by using the Solid Works 2017 and save it as *IGES format file* and export it to the ANSYS 16.1 software. Figure (3) represents the three-dimensional geometry of helical compression spring by ANSYS 16.1., while mesh generated as follows (*mesh size: proximity and curvature, relevance center: medium, span angle center: medium, and smoothing: medium*) as shown in figure (4).

Table 4: Geometry Parameters

Parameters	Value	Units
Wire Diameter	13	*mm*
Mean Diameter	127	*mm*
Outer Diameter	140	*mm*
Free Length	425	*mm*
Pitch	72	*mm*
Number of Active Turns	7	--------
Number of Total Turns	9	--------

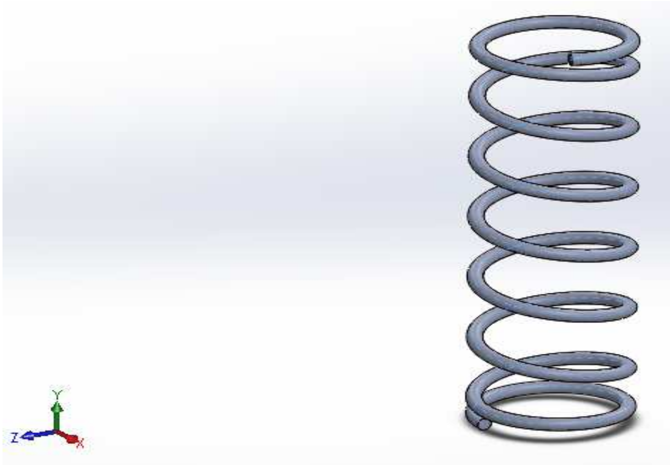

Figure 2: 3D Modelling of Helical Spring by Solid Works.

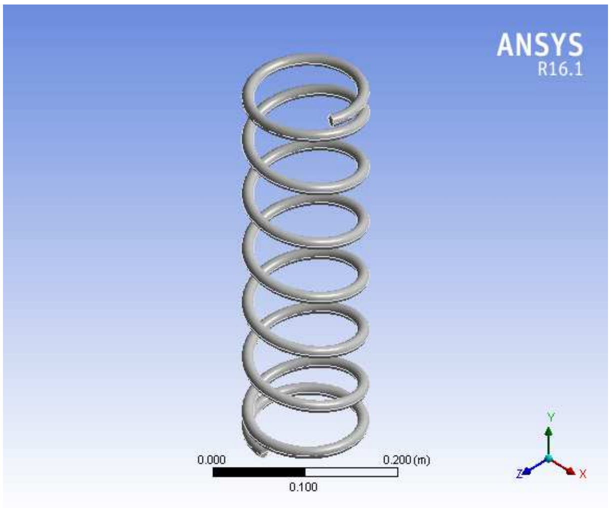

Figure 3: 3D Modelling of Helical Spring in ANSYS.

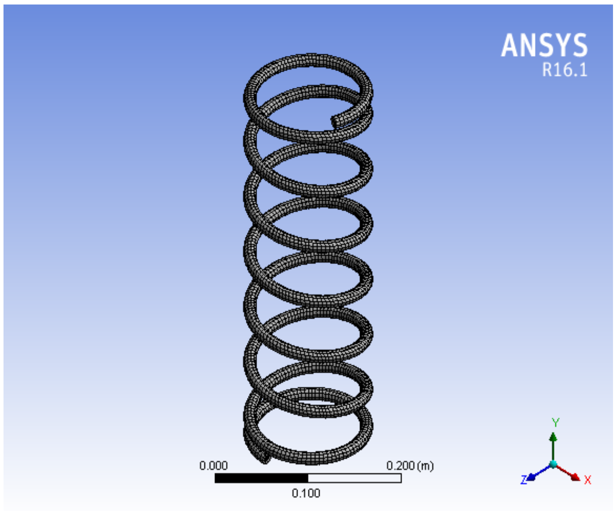

Figure 4: Mesh Generation of Helical Spring.

5. RESULTS AND DISCUSSIONS

The materials are examined by used two harmonic forces (4250–4500 N) as external forces affecting the helical spring. In addition, the harmonic frequency (at the sweeping phase: 0 degrees) as the range of (0 to 100 Hz) at an increased rate of (5 Hz). Through these forces and harmonic frequency, the deformation and shear stress distributions were calculated, and the frequency response of the two examined materials. Through these forces and harmonic frequency, the deformation and shear stress distributions were calculated as the frequency response of the two examined. The figures (5–8) illustrate the distribution of deformation, while the figures (9–12) show the distribution of shear stress. The figures (13–16) show the frequency response. Tables (5–7) explained the comparison of results obtained by numerical analysis of both deformations, shear stress and frequency response, respectively.

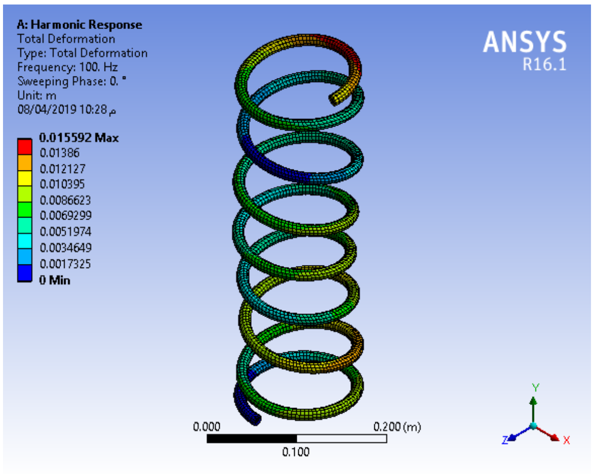

Figure 5: ASTM A227 Deformation at 4250 N.

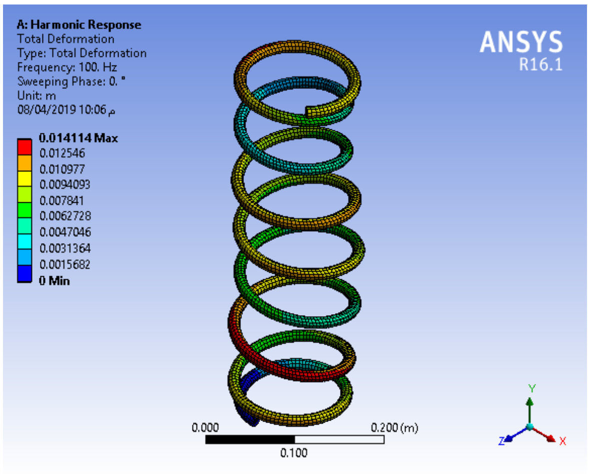

Figure 6: AISI 6150 Deformation at 4250 N.

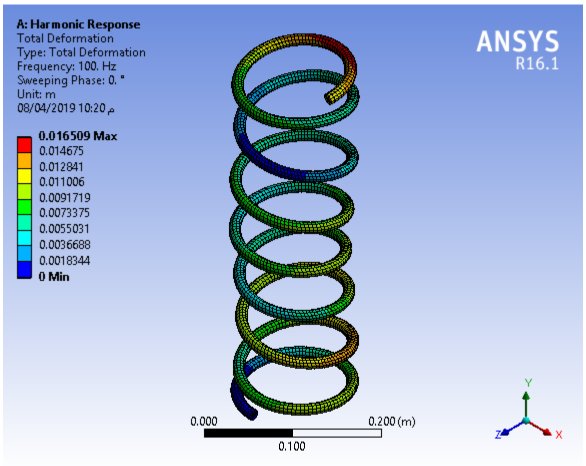

Figure 7: ASTM A227 Deformation at 4500 N.

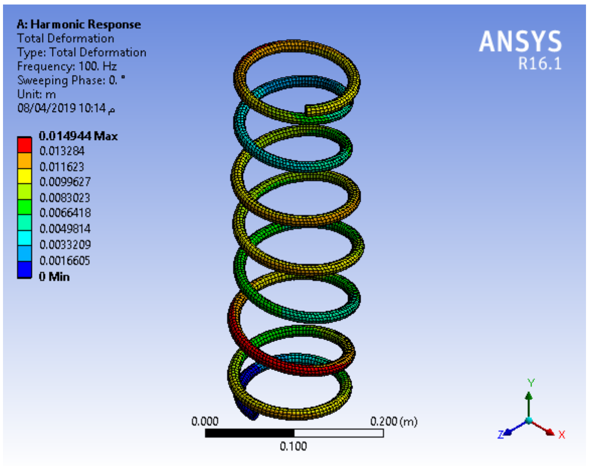

Figure 8: AISI 6150 Deformation at 4500 N.

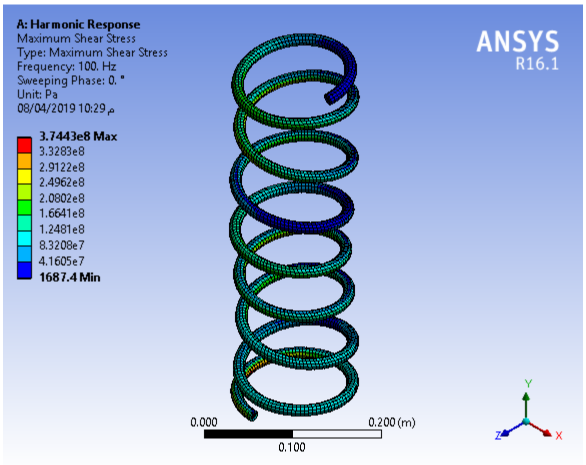

Figure 9: ASTM A227 Shear Stress at 4250 N.

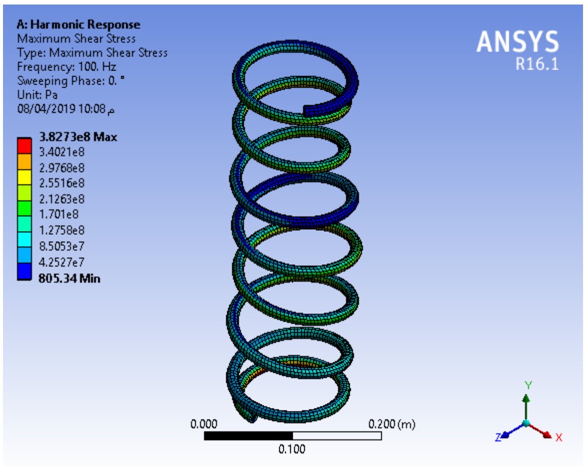

Figure 10: AISI 6150 Shear Stress at 4250 N.

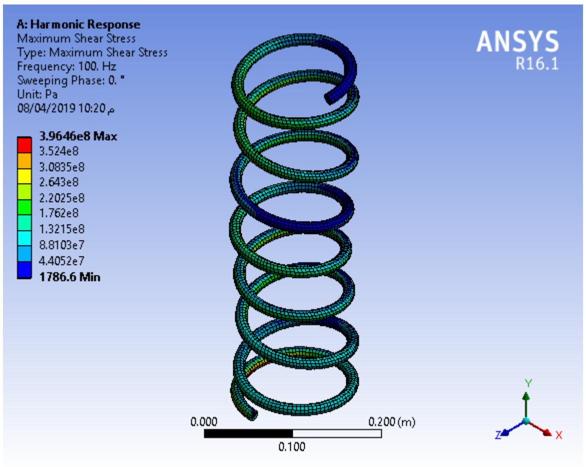

Figure 11: ASTM A227 Shear Stress at 4500 N.

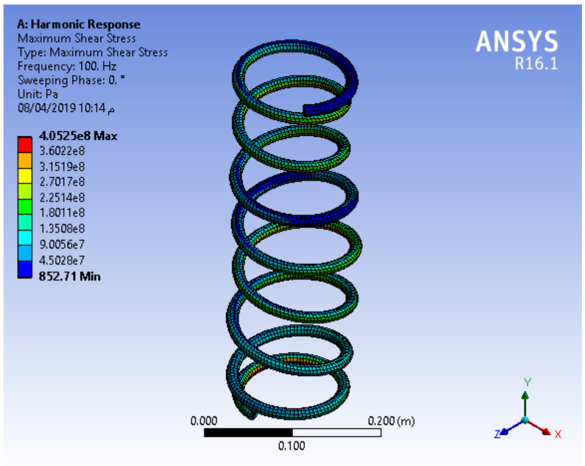

Figure 12: AISI 6150 Shear Stress at 4500 N.

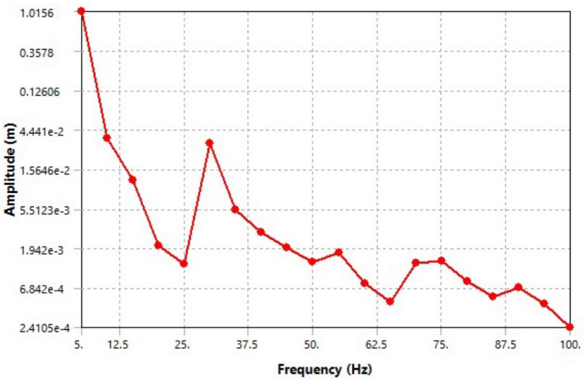

Figure 13: Frequency Rsponse of ASTM A227 at 4250 N.

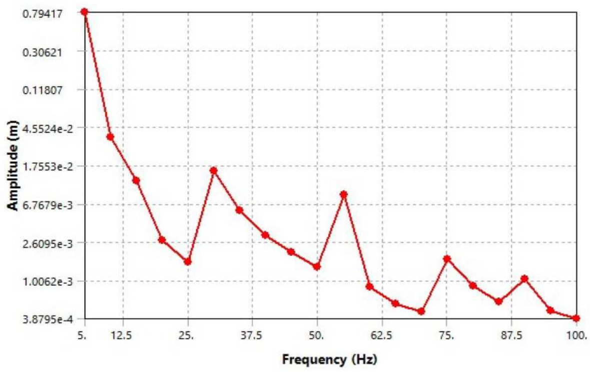

Figure 14: Frequency Response of AISI 6150 at 4250 N.

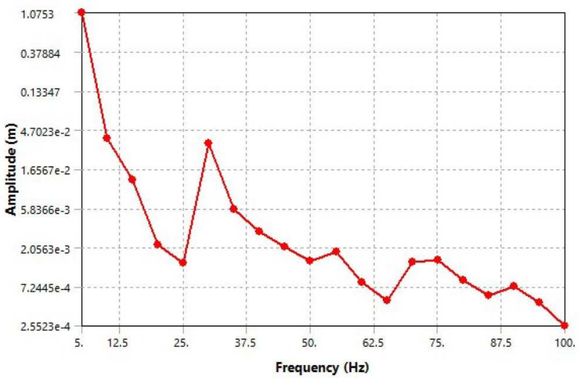

Figure 15: Frequency Response of ASTM A227 at 4500 N.

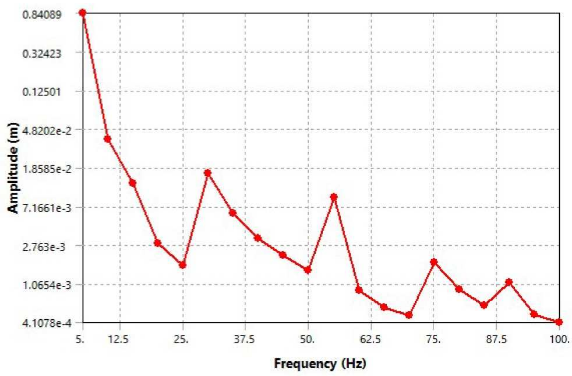

Figure 16: Frequency Response of AISI 6150 at 4500 N.

Table 5: Comparison of Deformation.

Force in *N*	Maximum Deformation in *mm*		Error percentage %
	ASTM A227	Chromium vanadium AISI 6150	
4250	15.592	14.114	9.4792
4500	16.509	14.944	9.4796

Table 6: Comparison of Shear Stress.

Force in *N*	Maximum shear stress in *MPa*		Error percentage %
	ASTM A227	Chromium vanadium AISI 6150	
4250	374.43	382.73	2.1686
4500	396.46	405.25	2.16903

Table 7: Comparison of Frequency Response

Force in N	Maximum amplitude in m		Error percentage %
	ASTM A227	Chromium vanadium AISI 6150	
4250	1.0156	0.79417	21.802
4500	1.0753	0.84089	21.79

6. CONCLUSIONS

Essentially, in the helical spring, it should resist the load which is subjected to its. However, the suspension system will be safe enough in the wheel. The numerical result of deformation, shear stress, and frequency response are compared between two selected materials. We get that numerical analysis by ANSYS, max. deformation (15.592, 14.114 mm) and max. shear stress (374.43, 382.73 MPa) and max. amplitude (1.0156, 0.79417 m) in ASTMA227 and Chromium vanadium AISI 6150 respectively at (4250 N) as affecting force. In addition, when using (4500 N) as an effective force, we observed increases in deformation (16.509, 14.944 mm) and shear stress (396.46, 405.25 MPa) as well as amplitude (1.0753, 0.84089 m) for same materials respectively. Percentage error between ASTM A227 and Chromium vanadium AISI 6150 of deformation is (9.4792 %–9.4796 %) and (2.1686 %–2.16903 %) for maximum shear stress and (21.802 %–21.79%) for the amplitude.

Finally, we have concluded that by using Chromium vanadium AISI 6150 would be highly durable and more efficient for use in suspension systems in the vehicles through riding on the slopes or bumped road.

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